

# Effects of Acoustic Barriers and Crosswind on the Operating Performance of Evaporative Cooling Tower Groups

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Most evaporative cooling towers are arranged on building roof due to the limitation of space and noise, and acoustic barriers are always installed around cooling towers in practical applications. The existence of acoustic barriers and crosswind may affect the recirculation phenomenon which is directly related to the operating performance of cooling towers. In this study, a physical and mathematical computation model is proposed to research the crosswind and distance between acoustic barriers and inlet of cooling towers. Both sensible and latent heat are considered in this research. The reflux flow rate and performance ratio are obtained to evaluate the recirculation and operating performance, respectively. The results show that the higher the crosswind velocity, the larger the reflux flow rate, and the lower the performance ratio of cooling tower groups. For high crosswind velocity, the presence of acoustic barriers is useful to inhibit reflux and improve operating performance, especially for *ICE* cooling tower groups. In addition, the optimum values are recommended for *LiBr/ICE* cooling tower groups in the research cases. The variation of reflux flow rate and performance ratio with the acoustic barriers' distance presents a parabolic tendency.

**Keywords:** evaporative cooling tower groups, acoustic barriers, crosswind, reflux flow, operating performance

## Introduction

The cooling system of power plant has a great impact on the capacity and operational efficiency of the power plant. Numerous works related to the cooling system of power plant have been reported in a wide variety of scientific literatures, particularly with regard to the indirect air-cooling (IDAC) system [1]. L.J. Yang et al. [2] carefully discussed the wind effect on the performance and characteristics of air-cooled condensers (ACCs) by experimental and numerical methods. G. Barigozzi et al. [3] optimized the waste-to-energy cogeneration plant to

reach the maximum net power output, considering the combined wet and dry units of the cooling system. Hongfang Gu [4] investigated the mechanism and optimization of wind-break structures for IDAC towers.

However, the high operating costs and the corresponding hit on plant efficiency associated with IDAC systems make it an unsatisfactory option [5]. The mechanical draft evaporative cooling towers become increasingly popular for power plant cooling systems owing to the higher heat exchange efficiency as well as the lower water consumption [6]. However, most cooling towers have to be arranged on building roofs where have

Nomenclature	Acronyms
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JIA Li: Professor

$A$	Area (m <sup>2</sup> )	$IDAC$	Indirect air-cooling
$c_p$	specific heat at constant pressure (J/kg K)	$LiBr$	Lithium bromide
$D$	Kinematic diffusivity (m <sup>2</sup> /s)	$ICE$	Internal combustion engine
$d$	Moisture content (kg(vapor)/kg(dry air))	$AB$	Acoustic barriers
$d'$	Moisture content of saturated air (kg(vapor)/kg(dry air))	$RH$	Relative humidity
$h$	Enthalpy (J/kg)	<b>Greek symbols</b>	
$h'$	Enthalpy of saturated air (J/kg)	$\varepsilon$	Turbulence dissipation rate (m <sup>2</sup> /s <sup>3</sup> )
$h_t$	The total enthalpy (J/kg)	$\varphi$	Evaporation loss rate (-)
$k$	turbulent kinetic energy (m <sup>2</sup> /s <sup>2</sup> )	$\gamma$	Volumetric porosity (-)
$Ka$	Mass transfer coefficient (kg/m <sup>3</sup> s)	$\eta$	Performance ratio (%)
$K_1, K_2$	Slope and intercept of the fitting relationship (-)	$\mu$	Dynamic viscosity (Pa s)
$L$	The distance between acoustic barriers and inlet of cooling tower (m)	$\rho$	Density (kg/m <sup>3</sup> )
$M$	Molecular weight (kg/mol)	<b>Subscripts</b>	
$p$	Pressure (Pa)	$a$	Air
$q$	Mass flow rate (kg/s)	$atm$	Atmosphere
$q_{v,a}$	Volume flow rate of air (m <sup>3</sup> /s)	$b1, b2$	Reflux flow for the group of cooling tower itself and the other surrounding one
$R$	Universal gas constant (J/mol K)	$c$	Cooling tower
$r$	Latent heat of vaporization (J/kg)	$f$	Process fluid
$S_S$	Source term of sensible heat (J/m <sup>3</sup> s)	$fr$	Fan region
$S_M$	Source term of latent heat (kg/s <sup>2</sup> )	$fl$	Filing layer
$t$	Temperature (°C)	$in$	Inlet
$t_w$	Temperature of spray water (°C)	$m$	Moist air
$t_{db}$	Dry-bulb temperature (°C)	$out$	Outlet
$U$	Velocity vectors (m/s)	$w$	Spray water
$V_{TB}$	Volume of tower body (m <sup>3</sup> )	$wv$	Water vapor
$v$	Velocity (m/s)		

limited land resources and some noise reduction facilities like acoustic barriers are needed [7]. The operating performance of mechanical draft evaporative cooling towers might be affected by these facilities, which have direct impact on the expected benefits of projects because of investment quota. It is important to study the layout plan of noise reduction facilities as well as the operating performance of cooling towers whether technical or economic reasons [8,9].

There are kinds of theoretical models of evaporative cooling process for both design and system simulation. Wojciech Zalewski and Piotr Antoni Gryglaszewski [10] developed a mathematical model of heat and mass transfer process in evaporative condensers which consists of four ordinary differential equations with their boundary conditions and some associated algebraic equations. A general non-dimensional mathematical model for description of all types of evaporative cooling devices was

presented by Boris Halasz [11]. He made the differential equations transformed to a pure non-dimensional form by the introduction of dimensionless coordinates and parameters. The description of the entire process becomes very simple. Sergey Anisimov and Pascal Stabat [12,13] adopted the e-NTU method to the indirect-contact evaporative cooling tower behavior and introduced only two parameters including air-side and water-side heat-transfer coefficients that allow one to estimate energy and water consumptions under different operating conditions. Based on theoretical research, numerous studies about nature or mechanical draft cooling tower were reported in scientific literatures [14-16].

Recently, some researchers studied about multiple cooling towers or cooling tower groups with the cross-wind and wind break wall. A. Chahine et al. [17] and Yuanshen Lu et al. [18] investigated the effect of cross-wind on the operating performance and plume dispersal

of cooling towers by the experiments and numerical simulation. They found that the crosswind sometimes may cause recirculation phenomenon and decrease operating performance at the same time. Z. Zhai et al. [19] and Y. Lu et al. [20] used wind break wall within the cooling towers which reverse the negative effect of crosswind.

But rare works in related to the impact on cooling tower groups with external walls or obstacles were reported which often occur in real cases. The phenomenon, such as low pressure, recirculation which means that the exit flow re-enters the inlet of cooling tower, and vortex flow generated around the cooling towers are always related with the cooling tower shape and the external walls or obstacles around towers as well as ambient conditions like crosswind. J.H. Lee et al. [21] considered the recirculation phenomenon and plume generation of a row of cross-flow type forced draft cooling towers surrounded by obstacles, and most of the results focused on the moisture fraction.

The present paper numerically studied the operating performance and recirculation phenomenon as well as the internal and external flow of two cooling tower groups surrounded by acoustic barriers with ventilating shutters in the lower position. Two main source terms representing the latent and sensible heat of each cooling tower were investigated, which are great meaningful for the discussion of operating parameters of cooling towers. The evaporation of spray water and sensible heat were obtained from the simulation results as well as the enthalpy and moisture content at the inlet of cooling towers. The effects of the distance between the acoustic barriers and cooling towers and the crosswind velocity were studied.

## Theoretical model

Two groups of cross-flow cooling towers were studied in this research. The water vertically fell from the upper part of the tower and the air horizontally flowed through the filler layers. The air and water flow were orthogonal in the cooling tower. Cross-flow cooling towers are widely used in the field of generation and air conditioning with many advantages, such as energy-saving, lower pressure and wind resistance, etc.. Besides, multiple tower can be placed on constructing basis freely built according to building shape and started with single or multiple tower according to the desired temperature [22].

## Heat and mass transfer process [10, 13, 23]

The heat and mass transfer process of cooling tower can be divided into two stages: heat transfer from the cooled process fluid (herein refers to water) inside the tube to the water film outside the tube, and mass diffusion process to promote heat transfer by the latent heat of vaporization from the water film to the forced convection air outside the tube. It is a complex coupling of heat and

mass transfer process [24,25]. The specific implementation process is that the heat transfer through the tube wall from the fluid inside the tube to the water film formed by the spray water outside of the tube. After that, heat and mass transfer between the humid air in the boundary layer near the water film and the main air flow of the cooling tower should be considered. Finally, the hot and humid air is driven into the atmosphere by the fan [26].

## Assumptions

In order to simplify the theoretical model, the following assumptions are assumed:

1. The main air of the cooling towers have the same state with the inlet air;
2. The temperature of the saturated humid air in the boundary layer near the water film is the same as the water film;
3. The inlet air has the same state with the ambient air when the cooling tower is in an ideal operating state at rated conditions (no crosswind, no acoustic barriers and no recirculation).

## Model of heat and mass transfer

Heat and mass transfer process for each cooling tower occurs between the main air and spray water outside the tubes. The total heat exchange of cooling towers including sensible heat and latent heat is simplified to act on the tower body portion shown in Fig. 2.

$$Q = Q_S + Q_L \quad (1)$$

Where,  $Q$  can be expressed as the heat loss of process fluid inside the tubes of cooling towers, because it is equal to the total heat exchange occurred between the main air and spray water outside the tubes.

$$Q = q_f \cdot c_{P_f} \cdot (t_{f,out} - t_{f,in}) \quad (2)$$

Where,  $q_f$  is the mass flow rate of the process fluid, and  $t_{f,out}$ ,  $t_{f,in}$  represent the outlet and inlet temperature of the process fluid, respectively.

The heat obtained by the air outside the tube, i.e., the sensible heat between the main air and the saturated air in the boundary is defined as:

$$Q_S = Ka \cdot (h' - h) \cdot V_{TB} \quad (3)$$

Where,  $Ka$  is the mass transfer coefficient between the water film and air, and  $h'$ ,  $h$  represent the enthalpy of saturated air in the boundary and main air, respectively.  $V_{TB}$  is the volume of tower body in the simplified model of cooling tower where heat and mass transfer occurs.

The heat loss of water film outside the tube which denotes the latent heat between the main air and the saturated air in the boundary reads as:

$$\begin{aligned} Q_L &= q_f \cdot c_{P_f} \cdot (t_{f,out} - t_{f,in}) - Ka \cdot (h' - h) \cdot V_{TB} \\ &= q_f \cdot \varphi \cdot r \end{aligned} \quad (4)$$

Where,  $\varphi$  and  $r$  represent the evaporation loss rate of cooling tower and latent heat, respectively.

According to the above assumptions and heat and mass transfer model, the mass transfer coefficient between the water film and air using the parameters of cooling tower under the rated conditions can be obtained, as shown in Table 1.

**Table 1** Rated conditions of each cooling tower

Parameter	LiBr cooling tower	ICE cooling tower
$t_{db}$	31	31
$RH$	56	56
$t_{f,in}$	38	95
$t_{f,out}$	32	74
$q_{v,a}$	755	184
$P_{fr}$	162.64	207
$\varphi$	0.988	3.44
$t_w$	33	35

## Numerical simulation

Two cooling tower groups surrounded by acoustic barriers with ventilating shutters in the lower position were numerically simulated and the standard  $k-\varepsilon$  model was selected. Constants of the turbulence model were given in Table 2. The distance between the acoustic barriers and cooling towers as well as the crosswind are taken into consideration.

**Table 2** Constants of standard  $k-\varepsilon$  turbulence model

$k$	$E$	$C_\mu$	$C_{\varepsilon 1}$	$C_{\varepsilon 2}$	$\sigma_k$	$\sigma_\varepsilon$
0.41	9.793	0.09	1.44	1.92	1.0	1.3

## Geometric model

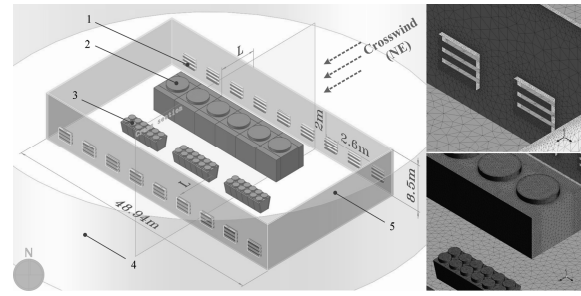
Fig. 1 illustrates the geometry and mesh configuration of 3D computational domain with acoustic barriers. The layout of cooling tower groups and acoustic barriers with ventilating shutters were given in Fig.1. The main geometric dimensions and crosswind direction also could be obtained. Seven different distances between the inlet of cooling tower and acoustic barriers from 1 m to 17 m were simulated and analyzed in this study. The crosswind velocities were of 0 m/s, 2.6 m/s, 6.6 m/s and 10.6 m/s with the wind direction at NE and the atmosphere temperature of 31 °C. Those cases with no acoustic barriers were also simulated for comparison.

## Governing equations

The continuity equation can be expressed as:

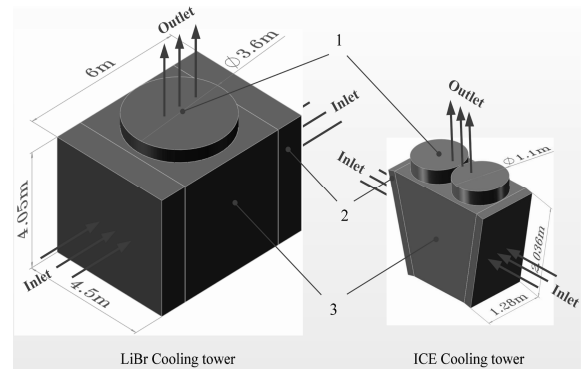
$$\nabla \cdot (\rho_m \mathbf{U}) = 0 \quad (5)$$

Where,  $\rho_m$  and  $\mathbf{U}$  represent the density and velocity vectors of humid air, respectively.



1. Ventilating shutter; 2. LiBr cooling tower group; 3. ICE cooling tower group; 4. Computational domain; 5. Acoustic barrier

**Fig. 1** Geometry and mesh configuration of computational domain with acoustic barriers



1. Fan region( $S_{fr}$ ); 2. Filling layer( $S_f$ ); 3. Tower body( $S_s$  &  $S_M$ )

**Fig. 2** Simplified model of cooling tower

The momentum equation is written as:

$$\nabla \cdot (\rho_m \mathbf{U} \times \mathbf{U}) = -\nabla p' + \nabla \cdot \boldsymbol{\tau} + S_M \quad (6)$$

Where,  $p'$  is the modified pressure and  $\boldsymbol{\tau}$  is the stress tensor, which is related to the strain rate by:

$$\boldsymbol{\tau} = \mu_{eff} (\nabla \mathbf{U} + (\nabla \mathbf{U})^T) \quad (7)$$

The term  $S_M$  represents the external forces;  $\mu_{eff}$  is the effective viscosity accounting for turbulence and  $\mu_t$  is turbulent viscosity. Their relationship is as follows:

$$\mu_{eff} = \mu + \mu_t \quad (8)$$

$$\mu_t = C_\mu \rho \frac{k^2}{\varepsilon} \quad (9)$$

Where,  $k$  and  $\varepsilon$  represent the turbulent kinetic energy per unit mass and turbulence dissipation rate, respectively. And  $C_\mu$  is the turbulence constant of standard  $k-\varepsilon$  model shown in Table 2.

The energy equation is:

$$\nabla \cdot (\rho \mathbf{U} h_t) = \nabla \cdot (\lambda \nabla T) + \nabla \cdot (\mathbf{U} \cdot \boldsymbol{\tau}) + \mathbf{U} \cdot S_M + S_E \quad (10)$$

$S_E$  is the source term of energy and  $h_t$  is the total enthalpy which is in related to the static enthalpy  $h(T, p)$  by:

$$h_t = h + \frac{1}{2} \mathbf{U}^2 \quad (11)$$

### Simplified model of cooling tower

Each cooling tower of two groups studied in this paper was operating at rated conditions, but the presence of obstacles and crosswind affected the air flow rate and air state at the inlet/outlet of cooling towers [20, 27-29]. Simplified model of cooling tower consisted of three parts including fan region, filling layer and tower body, as shown in Fig. 2. It also described the acting positions of source terms and the dimensions of *LiBr/ICE* cooling tower. Several source terms effected on every part of the cooling tower model were defined. Heat and mass transfer process for each cooling tower occurred in the tower body portion.

### The source term of sensible heat

The sensible heat exchange of cooling tower can be defined as a source term of energy equation acting on the tower body:

$$S_S = Ka \cdot (h' - h_{c,in}) \quad (12)$$

Where,  $h_{c,in}$  is the enthalpy at the inlet of cooling tower in related to the dry bulb temperature and moisture content of the inlet air, i.e.,  $h_{c,in} = f(t_{c,in}, d_{c,in})$ , which can be obtained by the following empirical formula [30]:

$$h_{c,in} = 1005t_{c,in} + d_{c,in} \cdot (2501000 + 1860t_{c,in}) \quad (13)$$

### The source term of latent heat

The latent heat exchange of cooling tower is equal to the spray water evaporation heat. An additional variable  $q_{wv}$  (scalar) acting on the transport equation is defined to express the evaporation rate of spray water which is solved on the basis of each unit cell. The scalar transport equation is as follows:

$$\nabla \cdot (\rho_m U q_{wv}) = \nabla \cdot (\rho_m D_{q_{wv}} \nabla q_{wv}) + S_{q_{wv}} \quad (14)$$

Where,  $D_{q_{wv}}$  denotes the kinematic diffusivity of water vapor and  $\rho_m$  is the density of humid air described by the following expression [21]:

$$\rho_m = \frac{p \cdot \left( \frac{1}{1+d_{c,in}} M_a + \frac{d_{c,in}}{1+d_{c,in}} M_w \right)}{R \cdot (t_{c,in} + 273.15)} \quad (15)$$

Where,  $R$  is the universal gas constant;  $M_a$ ,  $M_w$  represent the molecular weight of dry air and water, respectively.

The evaporation capacity of spray water is defined as a source term of scalar transport equation effect on the tower body:

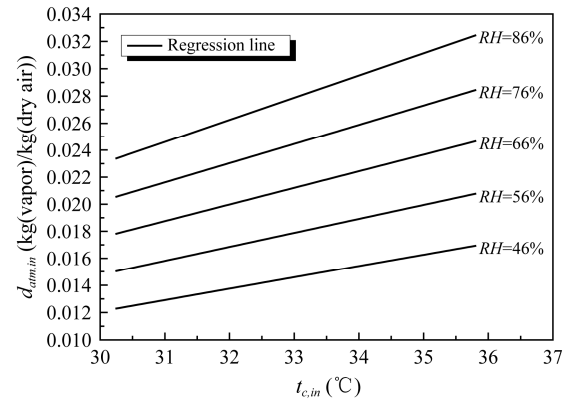
$$S_M = \frac{q_a \cdot [d' - d_{c,in}]}{V_{TB}} \cdot A_{out} \cdot v_{out} \quad (16)$$

Where,  $q_a$  is the mass flow rate of dry air;  $d'$  denotes the moisture content of saturated air near the water film;  $A_{out}$ ,  $v_{out}$  represent the area and mean velocity at the outlet

of cooling tower, respectively;  $d_{c,in}$  is the moisture content at the inlet of cooling tower obtained by the following expression:

$$d_{c,in} = d_{atm,in} + \frac{q_{wv,b1} + q_{wv,b2}}{q_a} \quad (17)$$

Where,  $q_{wv,b1}$ ,  $q_{wv,b2}$  represent the reflux flow of water vapor by the group of cooling tower itself and other surrounding groups, respectively.  $d_{atm,in}$  is the moisture content at the inlet of cooling tower having the same relative humidity with atmosphere, and it is a function of dry air temperature at the inlet of cooling tower and relative ambient humidity, i.e.,  $d_{atm,in} = f(t_{c,in}, RH_{atm})$ .



**Fig. 3** Moisture content versus dry air temperature at different relative humidity

The relationship of  $d_{atm,in} = f(t_{c,in}, RH_{atm})$  can be obtained using the database of Digital psychrometric Chart through which the moisture content data corresponding to the dry air temperature at different relative humidity is extracted and the data is fitted as shown in Fig. 3. The fitting relationship is defined as follows:

$$d_{atm,in} = K_1 t_{c,in} + K_2 \quad (18)$$

Where,  $K_1, K_2$  is the slope and intercept of the fitting relationship.

### The source term of filling layer

Considering the presence of fillers in a real cooling tower which will enhance the flow resistance and result in pressure loss and flow loss, the simplified model presented in this paper contains filling layer portions placed on both sides of the inlet of cooling tower. Filling layers can be regarded as porous medium using the square resistance coefficient of directional loss model to express:

$$S_{fl} = K_{loss} \frac{\rho_m}{2\gamma^2} \quad (19)$$

$$K_{loss} = \frac{\xi}{D_e} + \frac{\zeta}{L_e} \quad (20)$$

Where,  $\gamma$  denotes the volumetric porosity of the filling



layer;  $\zeta$  and  $\zeta$  represent the friction coefficient and the local resistance coefficient, respectively.  $D_e$  is equivalent diameter and  $L_e$  is the flow length.

### The source term of fan region [31]

Because the groups of mechanical draft cross-flow cooling tower studied were all driven by the fan at the top of tower body, fan model which could be set as a general momentum source term was introduced in the present paper:

$$S_{fr} = \frac{P_{fr}}{L_{fr}} \quad (21)$$

$$S_{fr,coef} = \frac{q_{v,a} \cdot \rho_m}{3600 \cdot V_{fr}} \quad (22)$$

Where,  $P_{fr}$  denotes the total pressure of fan and  $q_{v,a}$  is the volume flow rate of air through cooling tower;  $L_{fr}$  and  $V_{fr}$  are the height and volume of fan region, respectively.

### Boundary conditions

A cylindrical region was selected as a whole computational domain to represent the atmospheric condition. The height and diameter of the region were 18 m and 80 m, respectively. The crosswind velocity and temperature were used as the inlet boundary conditions defined on the side of the cylinder in the inflow direction. Opening boundary conditions acting on the other side and top of the cylinder as the outlet flow boundary were defined using the static pressure and temperature to describe the atmospheric conditions. Besides, solid lines in the following figures indicated these cases with acoustic barriers, and dash lines denoted the comparing cases without acoustic barriers.

### Results and analysis

In the present paper, seven different distances between the inlet of cooling tower and acoustic barriers of 1 m, 3 m, 5 m, 7 m, 9 m, 13 m and 17 m were investigated as well as the crosswind wind velocity of 0 m/s, 2.6 m/s, 6.6 m/s and 10.6 m/s. Furthermore, the cases without acoustic barriers at different crosswind velocities were also simulated as a comparison with the above. Total of 32 cases were studied and discussed in this paper. The mean parameters of two groups cooling tower and streamlines on a representative cross section were analyzed as follows.

#### Effect on the reflux flow rate

The scalar values at the inlet of cooling tower group could be obtained to represent the reflux flow rate of water vapor, and the evaporation rate of spray water have been defined as a scalar acting on transport equations for both *LiBr* and *ICE* cooling tower. The reflux flow consists of two parts, namely, their own reflux and the other

group's reflux. It can directly reflect the strength of recirculation phenomenon.

Fig. 4 shows the reflux flow rate in different cases for *LiBr* cooling tower group. Overall, the reflux flow rate increased with crosswind velocity whether or not there were acoustic barriers. Nevertheless, the increasing rate of reflux flow with  $AB$  decreased along with the rising of crosswind velocity in contrast with the cases without  $AB$  (Table 3). The reason is that the augmentation of crosswind velocity leads to the deterioration of flow conditions at the inlet and outlet of cooling towers and the existence of acoustic barriers sometimes may play a blocking role for the crosswind. For  $v_f=0$  m/s and  $v_f=2.6$  m/s, the reflux flow rate decreased with the augmentation of distance  $L$  and then remained nearly constant after  $L=9$  m. However, for  $v_f=6.6$  m/s and  $v_f=10.6$  m/s, the reflux flow rate increased with the distance after  $L=9$  m. A parabolic tendency with a minimum point was shown. This means that the reflux flow rate is larger than the cases without acoustic barriers under the same circumstances when acoustic barriers are close to the inlet of *LiBr* cooling tower. The limited surrounding space near the inlet of cooling towers will result in the reduction of air flow rate as well as the generation and change of vortex. Comparing to the case of  $v_f=10.6$  m/s without  $AB$ , the corresponding circumstance exhibited good phenomenon that acoustic barriers play a positive role in suppressing reflux.

The influence of the crosswind velocity on reflux flow rate for *ICE* cooling tower group (Fig. 5) was roughly consistent with the above-mentioned one. But the reflux

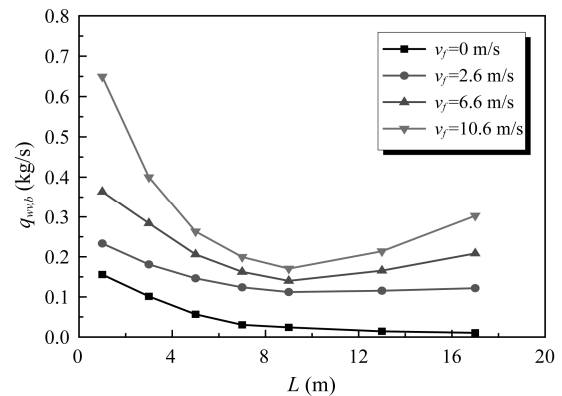
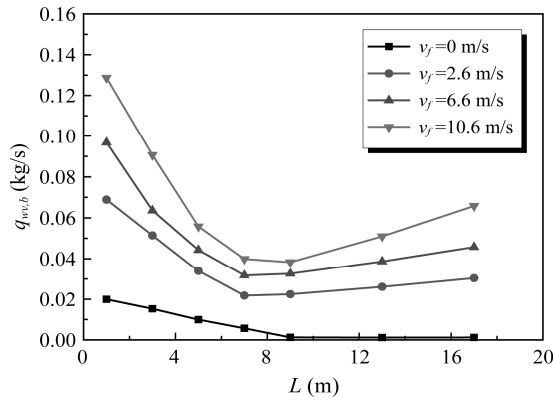


Fig. 4 Reflux flow rate versus the distance and crosswind velocity for *LiBr* cooling tower group with  $AB$

Table 3 Reflux flow rate for *LiBr/ICE* cooling tower group without  $AB$

	Reflux flow rate (kg/s)			
	0 m/s	2.6 m/s	6.6 m/s	10.6 m/s
<i>LiBr</i> cooling tower group	0.01023	0.08299	0.20599	0.46553
<i>ICE</i> cooling tower group	6.3356e-4	0.05885	0.09828	0.14524



**Fig. 5** Reflux flow rate versus the distance and crosswind velocity for *ICE* cooling tower group with *AB*

flow rate seemed to be more sensitive to acoustic barriers. In other words, the reflux flow rate significantly reduced for  $v_j$  varying from 2.6 m/s to 10.6 m/s when the distance of acoustic barriers is more than 1 m. The acoustic barriers played a positive role in these cases. In addition, the reflux flow rate decreased with the distance before  $L \leq 7$  m and then increased with  $L$ . The minimum reflux flow rate occurred at  $L=7$  m.

The operating performance of cooling towers would be directly affected by recirculation phenomenon caused by the suction at the inlet and the position of vortex and low pressure. On the one hand, the moisture content of the inlet air increased with the reflux flow rate when the air flow was constant. This means the rise of actual relative humidity at the inlet, which would deteriorate the evaporation rate of spray water and led to a sharp decline of sensible heat of cooling towers (Eq. (16), (17) and (18)). On the other hand, the reflux flow also resulted in the rising temperature of the inlet air. The decrease of sensible heat would be caused by the augmentation of enthalpy of main air due to the rising temperature and moisture content (Eq. (12), (13)), though the sensible heat accounted for only a small proportion of the total heat. In short, reflux flow is an unfavorable factor for the operating performance of cooling tower groups.

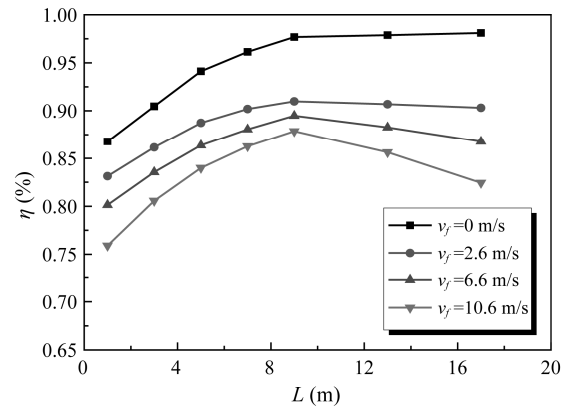
#### Effect on the performance ratio

It is not accurate and adequate to evaluate the operating performance of cooling tower groups simply by reflux flow rate. Some other impact factors, such as the air flow rate determined by acoustic barriers and crosswind, can also affect the evaporation of spray water (Eq. (17)). Hence, the parameter of performance ratio to represent the operating performance of cooling tower groups was suggested in this paper. It is defined as follows:

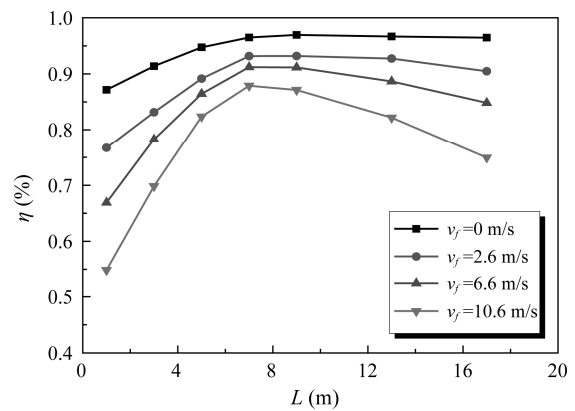
$$\eta = \frac{Q_{\text{real}}}{Q_{\text{rated}}} \quad (23)$$

Where,  $Q_{\text{rated}}$  is the rated total heat exchange of cooling tower groups under the rated conditions shown in Table 1;  $Q_{\text{real}}$  denotes the actual total heat exchange of those numerical cases, which can be obtained through the post processing.

Fig. 6 and Fig. 7 show the performance ratio of *LiBr* and *ICE* cooling tower groups, respectively. There are some similarities between the two figures. With the augmentation of the distance  $L$ , the performance ratios both increased at first and then decreased when the crosswind velocities varied from 2.6 m/s to 10.6 m/s. The transition points of the two cooling tower groups were occurred when the distances were 9 m and 7 m, respectively. As for  $v_j=0$  m/s, the performance ratios both increased with the distance  $L$  and then remained a constant after their transition point. The changes in acoustic barriers' distance obviously alter the flow field around cooling tower groups, due to the generation and changing position of vortex as well as the low pressure region caused by crosswind. When the crosswind did not exist, the suction of the cooling towers held a dominant position, and the



**Fig. 6** Performance ratio versus the distance and crosswind velocity for *LiBr* cooling tower group with *AB*



**Fig. 7** Performance ratio versus the distance and crosswind velocity for *ICE* cooling tower group with *AB*

performance ratios approximately remained a constant after transition point. In addition, the performance ratios both decreased with the rise of crosswind velocity no matter whether there were acoustic barriers or not. The decrease rates of the two groups both increased with the increase of crosswind velocity when the acoustic barriers did not exist.

But there are also some differences between the two figures. For example, the performance ratio of *LiBr* cooling tower group in most cases with *AB* was lower than the corresponding cases without *AB* for  $v_f=0$  m/s,  $v_f=2.6$  m/s and  $v_f=6.6$  m/s. The existence of acoustic barriers hinders the operating performance of cooling towers. However, contrary to the previous situation, acoustic barriers played a positive role in improving the operating performance of *ICE* cooling tower group compared to the corresponding cases without *AB* (Table 4). This indicates that it is easier for the *ICE* cooling tower group to be affected by crosswind. Moreover, the performance ratio was not completely in accordance with the reflux flow rate such as the comparison of cases for  $v_f=10.6$  m/s.

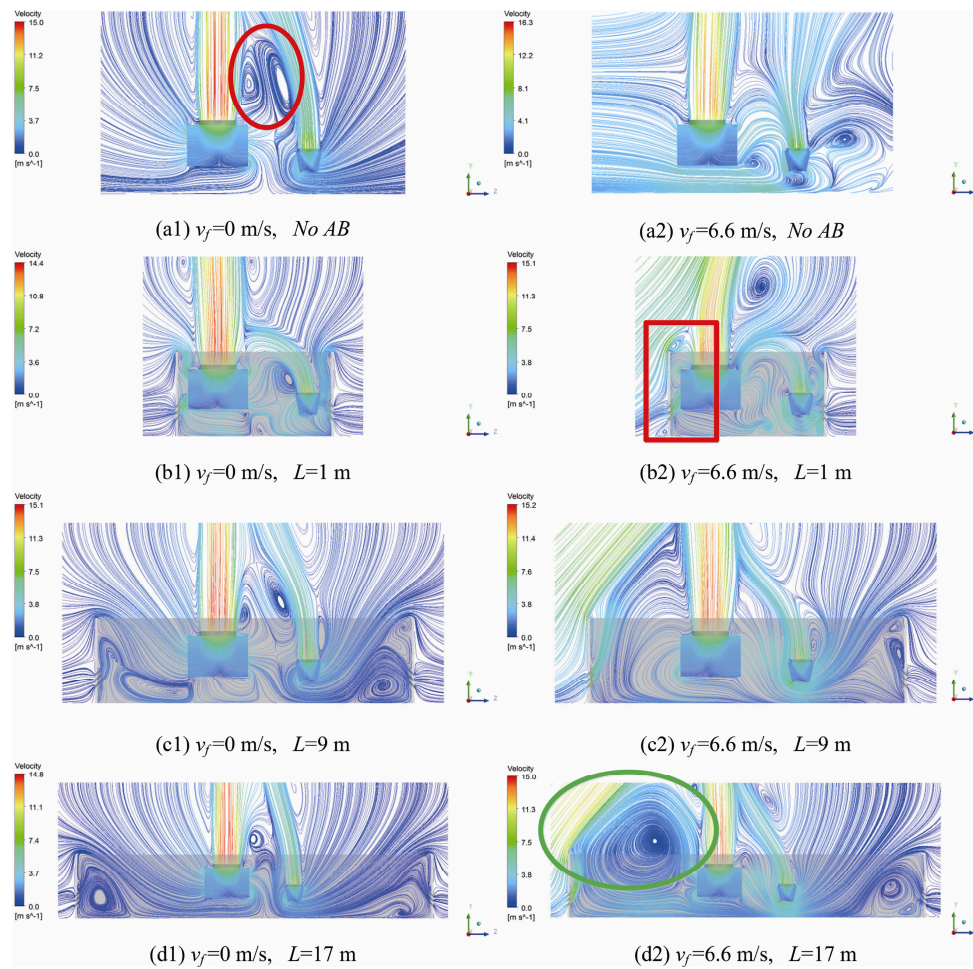
Fig. 8 displays the streamlines of different cases on a representative cross section shown in Fig. 1. It is visualized to describe the fluid flow situation around cooling towers through the streamlines. It is useful to explain the variant of operating performance qualitatively. Vortexes were generated in the low pressure region between the two cooling tower groups due to the high-speed flow at

the outlet shown in the red ellipse of Fig. 8 (a1). Comparing (b1), (b2) and (b3) of Fig. 8, it is obvious that the relatively superior flow field occurred when the distance  $L$  was 9 m. For  $L = 1$  m, there was not enough space around the inlet of cooling towers leading to the limitation of air flow rate. The space was enough for  $L = 17$  m, and low pressure region was formed when the crosswind blown over acoustic barriers. The vortex generated around cooling towers resulted in the rise of reflux flow and decrease of operating performance. In addition, it is easy to find that the existence of crosswind deteriorated the flow field for both the two cooling tower groups comparing the left and right sides of Fig. 8. *ICE* cooling tower group is more sensitive with crosswind due to the relatively small size. Even some tiny change can have a greater impact on it than the other one.

**Table 4** Performance ratio for *LiBr/ICE* cooling tower group without *AB*

	Performance ratio			
	0 m/s	2.6 m/s	6.6 m/s	10.6 m/s
<i>LiBr</i> cooling tower group	0.9839	0.9505	0.8834	0.7701
<i>ICE</i> cooling tower group	0.9775	0.8480	0.6787	0.4450





**Fig. 8** Streamlines on certain cross section at different crosswind velocity

## Conclusions

The recirculation phenomenon and operating performance of two groups cooling tower were investigated in the present study. The effects of crosswind velocities and distances between acoustic barriers and inlet of cooling towers were considered. The reflux flow rate and performance ratio were chosen as the evaluation criterion for recirculation and operating performance, respectively. Some results were obtained.

The recirculation and operating performance are strongly affected by crosswind velocity. The higher the crosswind velocity, the larger the reflux flow rate, and the lower the performance ratio of cooling tower groups, especially when there are no acoustic barriers.

The variation of reflux flow rate and performance ratio with the distance of acoustic barriers presents a parabolic trend. There are optimum values of the distance in this study for *LiBr/ICE* cooling tower groups appeared when  $L = 9$  m and  $L = 7$  m, respectively.

The presence of acoustic barriers plays a positive role for inhibiting reflux and improving operating perfor-

mance when the crosswind velocity is high, especially for *ICE* cooling tower groups.

The operating performance cannot reach the situation under the rated conditions even without crosswind and acoustic barriers. The vortices have been already generated between the two groups cooling tower.

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